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Performance investigation of linear compressor with one side springs

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ABSTRACT

In this paper a linear compressor prototype with a set of cylindrical coil springs connected to one side of the trestle is developed. The initial position of piston head is near to the top dead center and coil springs are on free status. The experiments on the prototype are carried out by compressing air on different working conditions. The experimental results show that the linear compressor with one side springs can take good advantage of the gas spring characteristics of the compressed work fluid to reduce the stiffness of mechanical spring and the weight of compressor. However the mechanical springs cannot help to constrain the piston bumping the cylinder head. The performance of the linear compressor prototype with one side springs changes with the discharge pressure sensitively. The discharge volume of the prototype on 0.2MPa is only 24.7% of that on 0.6MPa. The highest efficiency of the prototype is about 82.5% on discharge pressure of 0.5MPa and it drops with the decreasing of discharge pressure or discharge volume. On each condition with different discharge pressure or discharge volume, there exists the highest efficiency by changing the frequency. Therefore the linear compressor with one side springs is more suitable for the application that the cooling capacity need to be reduced with the decreasing of condensing temperature.

1. INTRODUCTION

Driven by a linear motor, the piston of linear compressor oscillates directly with the motor mover to compress the work fluid. Since the moving components of linear compressor are doing reciprocating motion, it is necessary to use mechanical spring as harmonic oscillator to resonate the inertia force, together with the gas spring generated by the compressed work fluid. Generally, there are two kinds of mechanical resonant spring used in linear compressor normally: planar spring and cylindrical coil spring. The linear compressors with planar spring are usually used for cryocooler or CPU cooling (R. Unger, 2002). Compare to planar springs, cylindrical coil springs are more economical and compact and are more suitable for the linear compressors used in civil field. LGE has developed the linear compressor with eight cylindrical coil springs and applied for refrigerator successfully for many years (H. Kim, 2009). The cylindrical coil springs are pre-compressed on the two sides of the trestle between the mover and the frame work. The displacement of the piston supported by two sides springs changes freely for its free piston mechanism and needs to be controlled accurately around the top dead center (TDC) to maintain high performance. The stiffness of the mechanical spring is designed much larger than that of the gas spring to reduce the change of the natural frequency on variable work conditions. In this way, when the linear compressor is developed for the applications with high gas force such as air conditioner, the quantities of the mechanical springs and the outline dimension of the linear compressor will be much larger because of high gas spring stiffness.

With the development of variable frequency technology, the power frequency can be adjusted timely with the change of natural frequency on different work conditions. Thus the stiffness of the mechanical spring need not to be larger than that of the gas spring and it is possible to save the material and contact the structure. The performance characteristics of the linear compressor with one side mechanical springs are investigated in this paper.

2. EXPERIMENTAL SETUP AND PROCEDURES

2.1 Experimental system

The experimental system takes air as the working fluid for it is convenient for the parameters measurement. Fig.1 is the schematic diagram of the experimental system. The linear compressor inhales the air (suction pressure: 0.1MPa)

from the atmosphere directly and discharges it back through the exhaust valve. The discharge pressure can be adjusted by the exhaust valve and measured by the pressure gauge. The variable frequency and voltage power supplies the power to the linear compressor and shows the parameters of frequency, input power, effective voltage, and effective current at the same time. The laser displacement sensor is installed on the framework of the linear compressor to measure the relative displacement of the piston to the framework. The data acquisition transfers the data from the sensors to the computer.

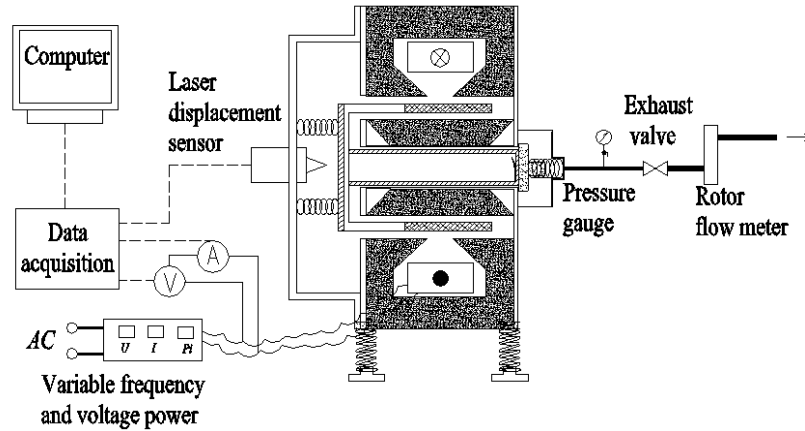


Figure 1: Schematic diagram of the experimental system

2.2 Configuration of linear compressor

Fig.2 shows linear compressor prototype with one side springs transformed from the Redlich type linear compressor. The initial position of the piston head is near to the TDC. Table1 shows the main parameters of the linear compressor prototype.



Figure 2: Linear compressor prototype

Table 1: The main parameters of the linear compressor prototype

Item	Prototype
Cylinder diameter, D (mm)	36.0
Mass of the mover, m (kg)	0.565
stiffness coefficient of mechanical springs, k_s (N/mm)	22.8

2.3 Experimental procedure

The experimental investigation is carried according to the following procedures:

(1) Measure the piston offset and discharge volume on different exhaust valve opening

Keep the exhaust valve on an opening degree, increase the discharge pressure from 0.2MPa to 0.6MPa step by step by increasing the voltage and record the piston displacement and discharge volume. Change the exhaust valve opening and redo the above steps.

(2) Compare the performance on the same exhaust valve opening and different discharge pressure

Set the exhaust valve on an opening degree, increase the frequency one by one from 47Hz to 60Hz and keep the discharge pressure on 0.2MPa by adjusting the voltage. Redo the same steps by changing the discharge pressure from 0.2MPa to 0.6MPa.

(3) Compare the performance on the same discharge pressure and different exhaust valve opening

Change the exhaust valve to another opening degree, increase the frequency one by one from 47Hz to 60Hz and keep the discharge pressure on 0.4MPa by adjusting the voltage.

3. EXPERIMENTAL RESULTS AND ANALYSE

3.1 Piston balance position offset

During the process of experiments, the position of the piston head changes with the opening of exhaust valve. The larger the exhaust valve opens, the closer the piston head get to the TDC. When the exhaust valve is opened too large, the piston will bump the cylinder head. The mechanical springs cannot constrain the piston bumping the cylinder head.

As the piston reciprocates to compress the air, the balance position of the piston movement will offset away from the initial position by the action of air force. The piston offset depends on the discharge pressure. In theory, the offset X_s are usually calculated as follows (G. Choe, 2000):

$$X_s = \frac{1}{2\pi k_s} \int_0^{2\pi} F_g(t) d(\omega t) \quad (1)$$

where k_s is the stiffness coefficient of mechanical springs, $F_g(t)$ is the gas force, ω is the power frequency.

Fig.3 is experimental and calculated results of the piston balance position offset on different piston stroke by changing the exhaust valve opening. The experimental results show that the piston offset is near to a constant with the increasing of piston stroke on the same discharge pressure and the higher the discharge pressure is, the larger the offset is. When the discharge pressure is set from 0.2MPa to 0.6MPa respectively, the offset changes from 1.58mm to 5.58mm. The calculated results show that the piston offset increases with the increasing of discharge pressure, while decreases with the increasing of piston stroke. However, it is noted that the calculated results are in well accordance with the experimental results at the TDC.

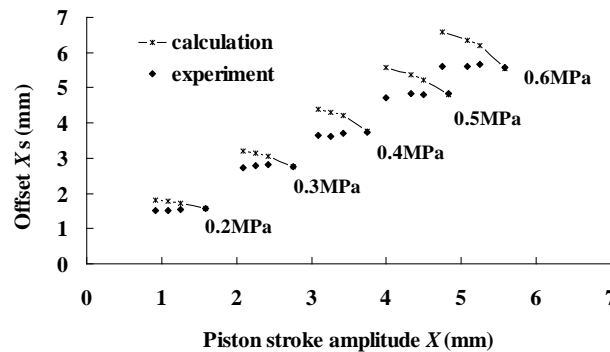


Figure 3: Offset on different piston stroke

Fig.4 is the experimental and calculated results of the discharge volume at TDC on different discharge pressures. The results between experiment and calculation are in well accordance. The errors are within $\pm 10\%$. The discharge volume changes with the discharge pressure dramatically. The discharge volume increases from 0.53 m³/h to 2.1m³/h with the increasing of the discharge pressure from 0.2MPa to 0.6MPa. The discharge volume on 0.2MPa is about 24.7% of that on 0.6MPa because of the piston balance position offset.

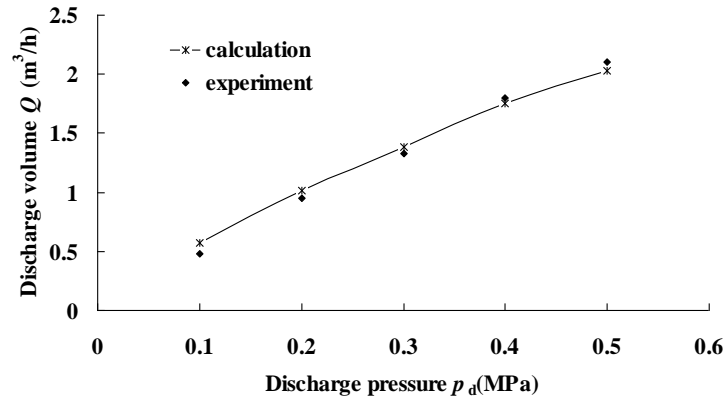


Figure 4: Offset on different discharge pressure

3.2 Performance investigation on different discharge pressure

The work cycle of the reciprocating compressor is generally regarded as a constant-quality isentropic process. Compression efficiency η is a significant parameter to show the performance of the linear compressor, which can be calculated as following:

$$\eta = \frac{P_o}{P_i} \quad (2)$$

$$P_o = f \int_0^T F_g(t) \frac{dx}{dt} dt \quad (3)$$

where P_i is the input power, P_o is the adiabatic compression work, $F_g(t)$ is the gas force (G. Choe, 2000), x is piston stroke, f is the power frequency.

Fig.5(a) shows the supply voltage curves adjusted to keep the same discharge pressure on different frequency. On the same frequency, the higher the discharge pressure is, the higher the voltage is needed. From 0.2MPa to 0.6MPa, all the voltage curves firstly decrease and then increase with the increasing of frequency. There is a critical frequency when the voltage is at the lowest value on different discharge pressure. The critical frequency decreases one by one from 56Hz on 0.2MPa to 52Hz on 0.6MPa.

Fig.5(b) shows the current curves change with the frequency. On the same frequency, the current increases with increasing of the discharge pressure. When the discharge pressure is set on a constant value, the current firstly decreases and then increases with the increasing of frequency from 47Hz to 60Hz. The critical frequency when the current is at the lowest value decreases one by one from 54Hz on 0.2MPa to 50Hz on 0.6MPa.

Fig.5(c) shows the input power curves change with the frequency. On the same frequency, the input power increases with the increasing of discharge pressure. The input power firstly changes little on the lower frequency and then increases obviously with the increasing of frequency. For example, the input power changes between 29.2W to 30.7W with the increasing of frequency from 47Hz to 55Hz and increases from 32.4W to 35.4W with the increasing of frequency from 56Hz to 59Hz on 0.3MPa, while changes between 122.0W to 123.1W with the increasing of frequency from 47Hz to 49Hz and increases from 129.0W to 146.4W with the increasing of frequency from 50Hz to 53Hz on 0.6MPa. The critical frequency decreases with the increasing of discharge pressure.

Fig.5(d) shows compression efficiency curves change with the frequency. When the discharge pressure is set on a constant value, the compression efficiency firstly increases and then decreases with the increasing of frequency. The higher the discharge pressure is, the faster the efficiency drops with the increasing of frequency. The highest compression efficiency of the prototype is 79.9% on 0.3MPa, 81.0% on 0.4MPa and 82.5% on 0.5MPa. The critical frequency when the compression efficiency is at the highest value decreases with the increasing of discharge pressure, that is 53Hz on 0.3MPa, 51Hz on 0.4MPa, 49Hz on 0.5MPa.

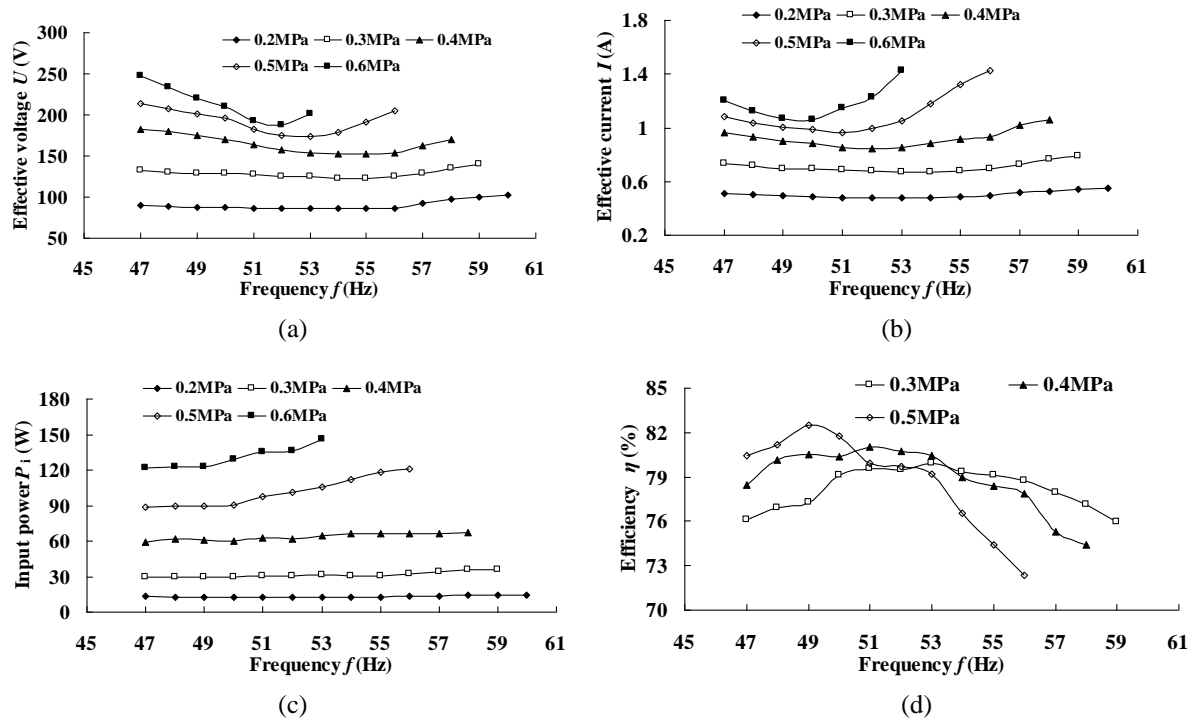
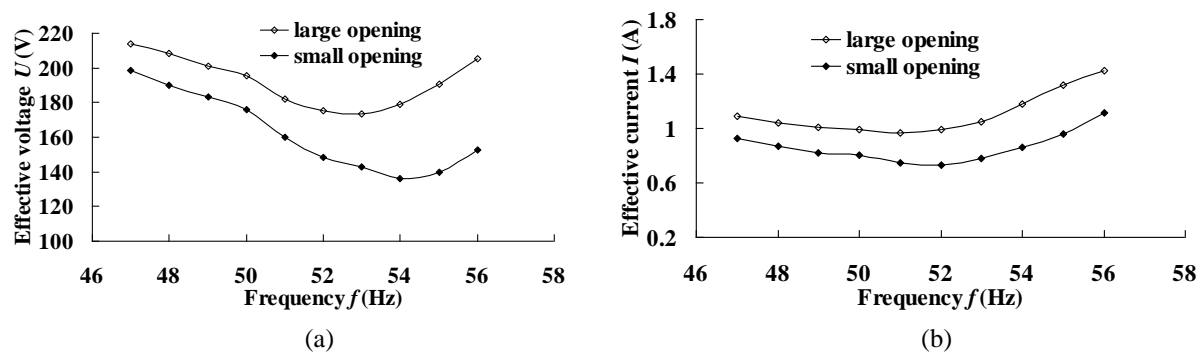


Figure 5: Relative parameters curves change with frequency on different discharge pressures (suction pressure: 0.1 MPa): (a) effective voltage, (b) effective current, (c) input power, (d) compression efficiency

The above experimental results show that on the same exhaust valve opening, all the supply power parameters of voltage, current and input power increase with the increasing of the discharge pressure. There exist the lowest value on the curves of effective voltage and effective current and the highest value on the curves of compression efficiency with the changing of frequency. The critical frequency decreases with the increasing of discharge pressure.

3.3 Performance comparison with different exhaust valve opening

Fig.6 shows the supply power parameters comparison on different exhaust valve opening on 0.4MPa. On the same frequency, effective voltage, effective current, input power and compression efficiency with large valve opening are higher than those with small valve opening. The lowest effective voltage is 173.6V on 52Hz with large valve opening and 136.1V on 54Hz with small valve opening. The lowest effective current is 0.966A on 51Hz with large valve opening and 0.732A on 52Hz with small valve opening. The input power changes between 89.1W to 90.8W with the increasing of frequency from 47Hz to 50Hz and increases from 97.6W to 120.8W with the increasing of frequency from 51Hz to 56Hz with large valve opening, while changes between 59.8W to 61.68W with the increasing of frequency from 47Hz to 51Hz and increases from 73.1W to 86.2W with the increasing of frequency from 52Hz to 56Hz with small valve opening. The highest compression efficiency is 82.5% on 49Hz with large valve opening and 79.9% on 50Hz with small valve opening.



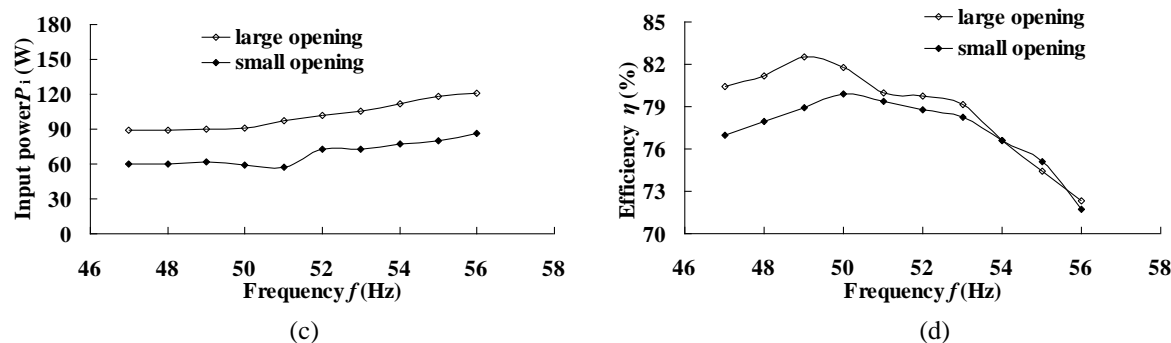


Figure 6: Relative parameters curves change with frequency on different valve opening (suction pressure: 0.1 MPa; discharge pressure: 0.5 MPa): (a) effective voltage, (b) effective current, (c) input power, (d) compression efficiency

The above experimental results show that the output of the linear compressor can be changed by adjusting the exhaust valve opening on the same discharge pressure. However, the compression efficiency will drop with the decreasing of output and there exists the highest compression efficiency by changing the power frequency on different conditions.

4. CONCLUSION

Linear compressor with one side springs can take good advantage of the gas spring characteristics of the compressed work fluid to reduce the stiffness of mechanical springs and the weight of the compressor. The above experimental results show that performance of the linear compressor prototype with one side springs changes with the discharge pressure sensitively.

- The piston balance position offset increases with the increasing of discharge pressure. On the same discharge pressure, the piston balance position offset is near to a constant on different discharge volume by changing the exhaust valve opening, while the position of the piston head gets closer to the TDC with the opening of exhaust valve. Since the mechanical springs cannot constrain the piston bumping the cylinder head and the piston will bump the cylinder head when the discharge pressure drops down rapidly with the opening of the exhaust valve, the piston stroke control system is still necessary to ensure its safe operation.
- Although the calculated results from theoretical formula are not in well accordance the experimental results on part output condition, the low errors between them at TDC show that the theoretical formula can be used to calculate the offset at TDC for in design of linear compressor one side springs design.
- The discharge volume changes obviously with the change of discharge pressure because of the piston balance position offset. The discharge volume of the prototype on 0.2 MPa is only 24.7% of that on 0.6 MPa. Therefore the linear compressor with one side springs is more suitable for the application that the cooling capacity need to be reduced with the decreasing of condensing temperature.
- On the same exhaust valve opening, the supply power parameters increase with the increasing of the discharge pressure. The output can be changed by adjusting the exhaust valve opening on the same discharge pressure, but the efficiency drops with decreasing of discharge volume. On each condition of different discharge pressure or exhaust valve opening, there exists the highest compression efficiency with the changing of frequency. The critical frequency decreases with the increasing of discharge pressure or the decreasing of output. According to these results, the highest efficiency of the linear compressor with one side springs on different work conditions can be got by adjusting the power frequency.

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